1 **Improvements in or relating to Vibration Control**

2

3 The present invention relates to improvements in or

4 relating to vibration control, and in particular to a

5 variable damper, a vibration control system incorporating

6 a variable damper and to a method of variably damping

7 relative motion between two members.

8

9 There are many situations where it is desirable to

10 control or damp the motion between two objects. One way

11 of doing so is to use a magnetorheological device, as

12 described for example in US 2,575,360. Magnetorheological

13 fluid (MRF) contains a suspension of paramagnetic

14 particles, such that when a magnetic field is applied,

15 the particles align with the field thus effectively

16 increasing the viscosity of the fluid.

17

18 A magnetorheological device typically contains an

19 electromagnet which generates a magnetic field when

20 current is passed through its coil. One moving part can

21 be enclosed within an MRF chamber such that when the

22 magnetic field is applied, there is opposition to

1 relative motion of that moving part with another moving

2 part.

3

4 US 5,492,312 describes a magnetorheological device

5 wherein a bolt and baffle plate assembly is contained

6 within an MRF chamber, the fluid in which can have a

7 magnetic field applied to oppose relative motion between

8 the assembly and an outer housing, thus damping motion in

9 up to six degrees of freedom. An electromagnetic coil is

10 formed around the outer periphery of the device.

11

12 However, design considerations have thus far limited the

13 application of magnetorheological devices for use in

14 devices where the forces that need to be controlled are

15 relatively high. For a device to support applications

16 such as those identified, the off-state force, namely the

17 minimum force required to induce relative motion between

18 the movable parts, needs to be low. This is difficult to

19 keep low because of the high density of the MRF, which

20 can only be reduced at the expense of its damping

21 effectiveness when a magnetic field is applied.

22

23 Furthermore an electromagnet can be variably controlled

24 such that the magnetorheological device provides varying

25 levels of damping. However, for such control to be

26 properly refined, there is a requirement that the forces

27 that would be expected to be applied to the device in use

28 fall within the force bandwidth, namely the off-state

29 force and the opposition force provided when the

30 electromagnet is fully activated.

31

It is an object of at least one embodiment of the present invention to provide a variable damper having a low or minimal off-state force.

It is a further object of at least one embodiment of the present invention to provide a vibration control system with improved damping.

According to a first aspect of the present invention, there is provided a variable damper comprising;

      an outer member including a magnetically conductive sleeve;

      an inner member comprising a shaft;

      an electromagnet supported between the members;

      wherein

      a chamber between the outer and inner members is at least partially filled with magnetorheological fluid (MRF), such that when a magnetic field is applied to the chamber, the effective viscosity of the fluid increases such that relative motion of the inner and outer members is opposed;

         the region between the electromagnet and the sleeve defining a control region in which the magnetic field is concentrated.

Preferably the outer member comprises a hollow cylindrical body having two body end surfaces, each in a plane perpendicular to the central axis of the shaft and spaced outwardly from an end of the electromagnet.

Preferably the sleeve is located against an inner surface of the outer member providing an inner sleeve surface

1 centred around the axis of the shaft and spaced outwardly

2 from the electromagnet.

3

4 Preferably, in a rest position in which no magnetic

5 field is applied, each body end surface is at a first

6 distance from an end of the electromagnet, and the inner

7 sleeve surface is at a second distance from the

8 electromagnet.

9

10 Alternatively the sleeve may have two sleeve end

11 surfaces, each in a plane perpendicular to the central

12 axis of the shaft and spaced outwardly from the

13 electromagnet.

14

15 The first and second distances represent variables that

16 define the size and shape of the control region. Here,

17 the distances as measured from the electromagnet are the

18 distances that are relevant. However, the electromagnet

19 may be encased within a housing, and the first and second

20 distances may be more conveniently defined as being the

21 distances between the housing and the surfaces.

22

23 Preferably, the first and/or second distances can be

24 minimised in order to reduce at least one degree of

25 freedom of the relative motion of the inner and outer

26 members.

27

28 In an embodiment, the electromagnet is supported on the

29 outer member. Preferably the electromagnet is supported

30 by a plurality of struts arranged perpendicular to the

31 shaft. These struts do not interrupt significantly the

32 flow path for the MRF through the control region. The

electromagnet is therefore fixed in relation to the outer member.

Bearings may be located between the electromagnet and the shaft for ease of manufacture these bearings will be inert.

In an alternative embodiment, the electromagnet is supported on the inner member.

Preferably, the inner member comprises interconnected first and second shaft portions, the longitudinal axes of which, when the inner and outer members are in a relative rest position, define the centre axis of the damper.

Preferably, a housing comprising the electromagnet is interposed between the first and second shaft portions.

Most preferably, a diaphragm seal portion is provided at each end of the shaft to bound the chamber.

Preferably, the shaft is magnetically inert. Thus only the electromagnet and sleeve need be magnetic. This allows the damper to be made primarily of light weight materials such as Aluminium and the like. Light weight materials decrease the off-state force.

Preferably, the seal portion has an elasticity to allow the inner member to rotate in planes perpendicular to the seal portion.

Optionally, the seal portion has an elasticity to reduce at least one degree of freedom of the relative motion of

the inner and outer members.

Preferably, the seal portion comprises a sprung collar
and a diaphragm seal.

Preferably, the device comprises an elastic end stop to
protect the device from damage induced from vibrations in
the case where the electromagnet fails.

The outer member may include a secondary housing at a
body end surface. The/each secondary housing may comprise
a hollow cylindrical body including an aperture through
which the shaft extends. Preferably plastic bearings are
located at the/each apertures against the shaft.
Preferably also, the/each secondary housing is filled
with a fluid such as air. These secondary housings act as
supports to assist in providing a single axis damper.

According to a second aspect of the present invention
there is provided a method of variably damping relative
motion between an outer member including a magnetically
conductive sleeve and an inner member, comprising the
steps:

    (a) supporting an electromagnet between the members
        such that a flow path exists between the
        electromagnet and the sleeve;

    (b) placing magnetorheological fluid between the
        members;

    (c) applying a minimal magnetic field to the
        electromagnet;

    (d) concentrating the field in the flow path; and

    (e) increasing viscosity of the fluid to thereby
        oppose relative motion of the membranes and
        create damping with minimal off-state.

According to a third aspect of the present invention,
there is provided a vibration control system for reducing
vibrations between a first and a second element, a
magnetorheological fluid variable damper being located
between the elements and operated to cause active damping
between the elements, wherein the system has a relative
figure of merit of less than 0.83.

The relative figure of merit provides an indication of
the reduction in vibration achieved by the system.

Preferably the relative figure of merit is equal to or
less than 0.5.

Preferably the magnetorheological fluid variable damper
is according to the first aspect.

Preferably the shaft is connected to the first element
and the housing is connected to the second element; and
the system further comprises a spring located between
elements; first and second accelerometers located between
the damper and the respective first and second elements;
and a control unit for inputting accelerometer values and
outputting a small electric current to the electromagnet,
to cause active damping between the first and second
elements.

Preferably the spring is a coil spring. Advantageously
the spring is one or more leaf springs. The leaf springs
may be arranged symmetrically around the damper. A 'c'
ring arrangement of one or more layered leaf springs may
be used. Leaf springs provide a controlled damping with

1   a reduction in dimensions of the vibration control

2   system.

3

4   Alternatively, the inner and outer members of the damper

5   are configured to be suitable for attachment to

6   components of each element on a device, such that the

7   application of relative forces between the elements

8   results in corresponding forces being applied to the

9   inner and outer members of the damper.

10

11   Preferably, a parasitic power generator is incorporated

12   within or on the device to provide the electric current

13   that drives the electromagnet.

14

15   Preferably the power generator comprises a plurality of

16   power generating units that are arrayed on the device at

17   points where concentrated load would be expected to be

18   applied to the device when it is put to use.

19

20   Preferably, the units comprise piezoceramic material.

21   Optionally, the units could comprise piezoelectric

22   unimorph or bimorph material.

23

24   Preferably, the device comprises at least one sensor

25   that detects a variable, the value of which can be

26   used to determine a desired amount of electric current to

27   be applied to the electromagnetic coil.

28

29   The current applied to the coil can be varied in order to

30   vary the strength of the magnetic field. In turn, the

31   effective increase in the viscosity of the MRF, and hence

32   the amount of damping between the inner and outer members

33   provided by the damper, is dependent on the strength of

1 the magnetic field. Thus, the desired amount of electric

2 current that is determined when a particular value of the

3 variable is detected can be representative of the desired

4 amount of damping that should be applied given that

5 value.

6

7 Preferably an intelligent control unit (ICU) is provided,

8 which is capable of receiving input signals from the

9 sensors and outputting command signals to the damper.

10

11 Preferably, an algorithm is used by ICU to determine

12 a desired relationship between the input signals and

13 the command signals.

14

15 Preferably, the device is a snowboard, one of the outer

16 member and inner member of the damper is attached to the

17 surface of the board, and the other of the inner member

18 and outer member is attached to a raised portion formed

19 on the board.

20

21 Preferably, the centre axis of the device is transversely

22 oriented with respect to the longitudinal axis of the

23 board.

24

25 Preferably, the centre axis of the device is parallel

26 with the longitudinal axis of the board.

27

28 Preferably, a plurality of dampers are attached to the

29 board. Dampers may be provided which have a mixture of

30 centre axis orientations as above.

31

1 Preferably, torsion forks are provided on the board and
2 connected to the inner member of the device to enable
3 control of torsional stiffness of the board.
4
5 Preferably, a piezoceramic power generating unit is
6 provided at a binding assembly.
7
8 The binding assembly is the point at which a boarder
9 would clip their boots into the board.
10
11 Optionally, the device is a golf club, one of the outer
12 member and inner member of the damper is attached to the
13 shaft of the club, and the other of the inner member and
14 outer member is attached to or forms the grip of the
15 club.
16
17 Optionally, the device is a handle which is a component
18 of a machine.
19
20 Such a "machine" may include, for example, a tennis
21 racket, polo mallet or other sports implement, or may be
22 a household tool such as a power drill, or may be a
23 bicycle or motorcycle, with the device being the
24 handlebar.
25
26 Optionally, the device is an engine mount, or pump mount.
27 Such devices are found in pumps, generators, engines,
28 vehicles and the like.
29
30 Embodiments of the present invention will now be
31 described, by way of example only, with reference to the
32 accompanying drawings, in which:
33

1 Figure 1 is an isometric view of a variable damper in
2 accordance with a first embodiment of the present
3 invention;
4
5 Figure 2 shows an isometric section of the damper shown
6 in Figure 1 illustrating the struts supporting the
7 electromagnet;
8
9 Figure 3 shows a sectional view of a variable damper in
10 accordance with a second embodiment of the present
11 invention;
12
13 Figure 4 shows a vibration control system according to an
14 embodiment of the present invention;
15
16 Figure 5 illustrates response curves for a vibration
17 control system with (a) a theoretical curve of
18 transmissibility against frequency, (b) an experimental
19 curve of transmissibility against frequency and (c) a
20 difference curve from which the relative figure of merit
21 for a system is derived.
22
23 Figure 6 shows a vibration control system according to a
24 further embodiment of the present invention;
25
26 Figures 7(a)-(d) show spring arrangements suitable for
27 use on a vibration control system according to an
28 embodiment of the present invention;
29
30 Figure 8 shows a snowboard incorporating two variable
31 dampers, according to an embodiment of the present
32 invention.
33

1 Figure 9 shows a partial cross sectional view of a

2 variable damper in accordance with a third embodiment of

3 the present invention, mounted longitudinally on a

4 snowboard;

5

6 Figure 10 shows a control schematic for the damper as

7 illustrated in Figure 9; and

8

9 Figure 11 illustrates a fourth embodiment of a variable

10 damper, as applied for use with a golf club.

11

12 Referring initially to Figure 1 of the drawings, there is

13 illustrated a variable damper, generally indicated by

14 reference numeral 10, according to a first embodiment of

15 the present invention. Damper 10 comprises an outer

16 cylinder 12 and an inner shaft 14. The inner shaft runs

17 symmetrically through the outer cylinder 12. Located

18 within the outer cylinder, against its inner surface 13

19 there is a sleeve 15. Steel sleeve 15 provides a magnetic

20 flux return guide within the damper 10.

21

22 Located within the steel sleeve 15 but independent of the

23 sleeve and of the shaft 14 is electromagnet 20.

24 Electromagnet 20 has a core 21 surrounded by a mount or

25 housing 23, which is magnetically inert. The core housing

26 23 is supported within the region between the shaft 14

27 and the sleeve 15 by mounts (not shown) onto the outer

28 cylinder 12. Thus a gap or flow path 25 exists between

29 the shaft 14 and the housing 23, and a flow path 27

30 exists between the core 21 and the sleeve 15.

31

32 The outer cylinder 12 defines a chamber 29 located

33 therein which is bounded at an upper 31 and a lower 33

1 end by diaphragm seals 35 (only one is shown at the lower

2 end 33). Each diaphragm seal 35 provides a solid collar

3 around the input shaft 14 which extends from the chamber

4 through the diaphragms 35 at either end. It is noted that

5 the shaft 14 is not connected to the EM core 21 although

6 in further embodiments it may be guided with respect to

7 the core, by the incorporation of bearings.

8

9 In this embodiment of the present invention, in which

10 diaphragm seals are provided as part of a piston. The

11 seals are connected by the input shaft which runs through

12 the electromagnet.

13

14 In use, a control volume 27 of MR fluid is constant

15 between the fixed electromagnet (EM) core 21 and the

16 magnetic flux return guide, sleeve 15. The electromagnet

17 20 is fixed inside the outer cylinder 12 - mounted inside

18 the steel sleeve 15 that acts as the magnetic flux return

19 guide. The input shaft 14 for connection to the vibration

20 source runs through the centre of the EM core 21, with

21 opposing diaphragms 35 connected to the shaft 14 and

22 sealing the system 10. Movement of the input shaft 14

23 relative to a fixed outer cylinder 12 (connected to the

24 structure to be damped against) results in a pressure

25 change in the MR fluid chamber 29 - driving the fluid

26 around the fixed EM core 20, in the annular orifice 27

27 between the core 20 and the sleeve 15.

28

29 Activation of the electromagnet 20 controls the flow of

30 the MR fluid around the electromagnet 20. Increasing

31 power to the electromagnet 20 results in an increase in

32 apparent viscosity of the MR fluid between the EM core 21

33 and sleeve 15. Exposing the control volume 27 of MR fluid

1   to a variable strength magnetic field enables the control

2   volume 27 to act as a flow control valve. Increasing

3   resistance to fluid flow enables the device to absorb

4   more energy from vibration induced movement of the input

5   shaft 14 relative to the outer cylinder 12.

6

7   Connecting the input shaft 14 to opposing diaphragms 35

8   (with a solid collar around the input shaft 14 at either

9   end, to act as a piston) ensures pressure induced by

10   movement of the input shaft 14 is equal in both

11   directions (i.e., up and down when considering figure 1).

12

13   The movement of fluid from regions experiencing

14   relatively small magnetic field into the control region

15   27 helps to reduce degradation in the performance of the

16   fluid (i.e., as a result of in-use-thickening).

17

18   One primary and two secondary degrees of freedom can

19   be controlled with the connected diaphragm actuator 10.

20   The primary degree of freedom is with the input shaft 14

21   reciprocating relative to the outer cylinder 12 (i.e., up

22   and down when considering figure 1). Additionally, pitch

23   and yaw about the common central axis of this axis-

24   symmetric actuator can be controlled (i.e., limited

25   rotational movement about two axes orthogonal to the

26   common central axis). This is largely possible due to

27   specification of a diaphragm seal 35, which is a

28   fundamental part of the piston that induces pressure

29   driven flow of the MR fluid around the EM core 20.

30

31   The input shaft 14 runs through the electromagnet core

32   21, but is not connected to it. To achieve control in

33   three degrees of freedom the input shaft is machined from

1   a magnetically inert material, so that its movement is

2   not influenced by the electromagnet.

3

4   Control of movement of the input shaft 14 relative to the

5   outer cylinder may be advantageous. This can be achieved

6   by guiding the input shaft through the EM core. A sprung

7   collar/bush between the outside diameter of the input

8   shaft and the inside diameter of the EM core can be

9   specified to control movement of the input shaft against

10 the EM core (i.e., lateral movement when considering

11 figure 1).

12

13 Additionally, damper mounts located outside on the outer

14 cylinder may be made from rubber and specified to act as

15 an end-stop to prevent movement of the structure

16 connected to the input shaft against the structure to

17 which the outer cylinder is fixed. Therefore, rubber

18 damper mounts around the outer cylinder can act as a

19 mechanical failsafe, should the electromagnet fail. Due

20 to the damage that may be caused should a vibration

21 control system fail, such a mechanical failsafe should be

22 considered a necessity in a number of applications of the

23 device.

24

25 By providing a magnetic field across only a small region

26 27 of the chamber 29, the damper can be operated from a

27 low off-state. This off-state may be typically 10-20N.

28

29 Figure 2 illustrates the damper of Figure 1 showing the

30 support means 41 used to locate the electromagnet 20

31 between the shaft 14 and the cylinder 12. The support

32 means comprises longitudinally arranged struts 41 which

33 connect the cylinder 12 to the housing 23 of the

1 electromagnet 20. These struts 41 merely provide support

2 and do not interrupt MR fluid flow through the region 27

3 or the return magnetic flux path.

4

5 A yet further feature of the damper 10 is illustrated in

6 Figure 2. In this embodiment, the housing or outer

7 cylinder 12 is extended at each end 31,33 respectively to

8 provide additional supports 43,45. Upper support 43 is a

9 thickening of the end face of the cylinder 12 and is

10 located over the diaphragm seal 35. Thus MR fluid in the

11 chamber 29 is held in, initially by the seal 35 acting on

12 the piston washer 47, and further by the bearing 49 on

13 the support 43. The bearing 49 is plastic and provides a

14 sliding fit against the shaft 14. An identical

15 arrangement is symmetrically arranged at the lower end 33

16 of the cylinder 12.

17

18 Figure 2 also shows an additional feature which may be

19 incorporated into the damper 10. Airflow channels 37 are

20 machined through the outer cylinder 12 to provide a

21 coolant flow path through the damper 10. In the

22 illustration there are four channels 37 shown, however it

23 will be appreciated that any number can be used, space

24 permitting.

25

26 The additional supports 43,45 force the damper 10 to

27 operate on a single axis ie the shaft 14 can only

28 reciprocate on the central axis through the damper 10.

29 This restricted movement of the shaft 14 provides the

30 damper with a low off-state when coupled with the short

31 control region 27.

32

33 Reference is now made to Figure 3 of the drawings which

1   illustrates a second embodiment of the present invention.

2   Like parts of those of Figures 1 and 2 have typically

3   been given the same reference numeral with the addition

4   of 100. The variable damper (also called an "actuator" or

5   an "MRF device ") 110 comprises an outer portion 112 and

6   an inner portion 114. The inner portion 14 comprises a

7   first portion 116, second portion 118, and an

8   electromagnet 120. Power lines 122 are provided within

9   the first portion 116 to power the coil 124 of the

10 electromagnet 120.

11

12 The first 116 and second 118 portions have seals 115,

13 which, together with an inner surface of the outer

14 portion 112 define an MRF chamber 128. When electric

15 current flows through the electromagnet coil 124, a

16 magnetic field 128 is induced, which has the effect of

17 increasing the effective viscosity of the MRF in the

18 chamber 128, the increase being dependent on the

19 power of current being passed through the coil 124.

20

21 Inner seals 115 and outer seals 117 together define the

22 seal portion of the inner member 14. Any suitable form of

23 seal may be used, suitably a diaphragm grommet seal.

24 Additionally the seal portion could be provided by a

25 sprung collar and diaphragm seal at opposite ends of the

26 inner portion, in a similar arrangement to the diaphragm

27 seal 35 and washer 49 of Figure 2.

28

29 Insertion of a sprung collar between the inner axle 114

30 and outer cylinder 112 provides resistance to movement,

31 proportional to the stiffness of the spring in a

32 particular axis. The MR fluid, electromagnet and sleeve

33 (or cylinder) adds control of dynamic movement.

1 The sprung collar provides primary control in two axes

2 orthogonal to the central axis and secondary control

3 along the central axis.

4

5 In an alternative embodiment, the sprung collar may be

6 replaced by a sprung bush, as is known in the art.

7

8 In further alternative embodiments which are not

9 illustrated herein, the sprung collar may have a

10 rectangular or square cross-section.

11

12 The incorporation of sprung collars or bushes between the

13 inner axle and outer cylinder has a number of benefits in

14 many applications, including;

15

16 1. To resist deflection of the inner relative to the

17   outer up to a specified off state force.

18

19 2. To return the inner and outer to their neutral, rest

20   separation.

21

22 3. To ensure the inner and outer do not actually touch.

23

24 4. To control axial movement.

25

26 Items 1 and 3 are conflicting requirements, so therefore,

27 a mechanical end-stop may be additionally specified (to

28 prevent the inner touching the outer), should it not be

29 possible with the same spring to provide a low off state

30 force and ensure clearance is maintained.

31

32 The spring constant does not necessarily have to be equal

33 at either end of the cylinder. This presents the

1 opportunity to control axial movement, with resistance to

2 movement (between the inner axle and outer cylinder) at

3 one end of the MRF device being greater or less than the

4 resistance at the other end.

5

6 The damper 110 operates in a similar manner to damper 10.

7 Controlling the viscosity of the MRF means that the

8 damping of relative motion between inner and outer

9 portions 12, 14 can be controlled.

10

11 A steel (or other magnetically conductive material)

12 sleeve 130 is mounted internally in the outer portion

13 112, which provides a flux return path through the

14 electromagnet 120 for the magnetic field. This has the

15 effect of concentrating the magnetic field in a region

16 132 between the inner and outer portions 112, 114,

17 defining a control volume of MRF within the chamber 128

18 that acts as a control region. It is the variation of the

19 viscosity of this control volume that is critical to

20 controlling the damping. MRF in the remaining volume of

21 the chamber 128 is not activated by the magnetic field

22 when it is applied.

23

24 The chamber 128 is bounded by the outer member 112,

25 rather than the sleeve 130. Thus, the volume of the MRF

26 in the device is larger than the control volume.

27

28 This ensures that fluid in the control volume can be

29 recycled with fresh fluid as the inner member 114 is

30 moved relative to the outer member 112, the MRF in the

31 control volume being moveable away from the electromagnet

32 to a region of the MRF chamber that is substantially

33 outside the magnetic field. This re-circulation of the

1   fluid reduces the likelihood of fluid-particle separation

2   and in-use thickening, to improve the longevity of the

3   device.

4

5   The housing that includes the sleeve 130 can be made from

6   a single component, where the outer housing is made from

7   steel and provides the field return path or up to three

8   components, where the steel sleeve 30 is assembled

9   between split cylinder that makes the outer housing.

10

11   This simple construction reduces the number of moving

12   components, making the damper 110 easy to manufacture,

13   and also making it durable.

14

15   The electromagnet 120 comprises copper wire wound around

16   a steel core mounted on an inner axle. Therefore, the

17   magnetic flux generator is axis-symmetrically mounted

18   with the MR fluid between it and an outer cylinder to

19   which a steel (or other magnetically conducting material)

20   cylinder is internally mounted - providing a flux return

21   path (to the electromagnet, through the MR fluid).

22

23   Mounting the electromagnet on the axle has been

24   considered the most power efficient means of generating

25   magnetic field in the system. Prior devices, in which a

26   coil is wound around the outer cylinder with a

27   magnetically conductive piston mounted on the axle to

28   complete the magnetic circuit, require considerably more

29   power in order to generate a comparable magnetic field

30   with the device thus constructed. However, the

31   arrangement of the electromagnet independently suspended

32   between the axle and the outer cylinder, providing a

33   control region between the electromagnet and the outer

1 cylinder, has been found by the inventors to give a more

2 power efficient means and thus a damper with the lowest

3 possible off-state force. This is the embodiment shown in

4 Figures 1 and 2.

5

6 The advantage of the embodiment shown in Figure 3 is that

7 it provides for multi-axis control. Two translational

8 degrees of freedom are provided, as the inner portion 114

9 translates in a direction along an axis running from left

10 to right of the device 110, or in a direction along an

11 axis extending normal to the page, as illustrated in

12 Figure 3.

13

14 When a magnetic field is applied, the resistance to this

15 relative translational motion that is provided by the MRF

16 is known as a pressure driven flow mode.

17

18 Activation of an electromagnet produces an apparent

19 change in viscosity in MRF exposed to the generated

20 magnetic field. As the MR fluid becomes more viscous,

21 more force is required to generate a pressure that causes

22 the fluid to flow around a constriction. The movement of

23 the electromagnet on the inner axle relative to the outer

24 steel sleeve creates a constriction and (pressure driven)

25 fluid flow can be controlled (like a valve) as the

26 electromagnet activates the MR fluid.

27

28 There is also a rotational degree of freedom for relative

29 rotation about a central axis 134 of the device 110.

30

31 When the two portions 112, 114 attempt to rotate relative

32 to each other in this way, the MRF resists the movement

33 by a shear force that is induced at the surfaces of the

1 chamber 128.This can be known as the direct shear mode of

2 damping control. The strength of resistance to motion

3 offered by the direct shear mode is much less than the

4 strength offered by the pressure driven flow mode.

5

6 The inner member 114 comprises a first portion 16, a

7 second portion 118, and a housing containing the

8 electromagnet 120. These portions are integral, and the

9 longitudinal axes of the first and second portions are

10 in-line and define a central axis 34 both of the inner

11 member 114 and the device 10.

12

13 Seals 115, 117 provide sufficient elasticity for the

14 shaft of the inner member to rotate about an axis running

15 into and out of the page of the device as illustrated in

16 Figure 3 (i.e. moving clockwise/anticlockwise in the

17 figure), and about an axis running from left to right

18 horizontally as illustrated in Figure 3 (i.e. tilting

19 into and out of the page in the figure).

20

21 Movement between the inner 114 and outer 116 portions in

22 a direction along the central axis 134 of the device is

23 limited in its extent by the seals.

24

25 Spring return to the neutral position results from the

26 viscoelastic property of the seals / with sprung collars

27 located between seals, against the inner and outer (i.e.,

28 in the space between the shaft and the outer housing).

29 Thus, the two translations at right angles to the shared

30 central axis (of the inner axle and outer cylinder), plus

31 pitch and yaw about the same axis can be considered as

32 being four primary degrees of freedom that can be

33 controlled, while one translation of the inner member

1 relative to the outer member along the shared axis and

2 one rotation about the same axis (assuming the diaphragm

3 seal is assembled to rotate with the inner axle) can be

4 considered as two secondary degree of freedom can be

5 controlled. The secondary degrees of freedom are limited

6 by the seals.

7

8 One advantage of the damper described above lies in its

9 ability to provide control of dynamic movement over a

10 range (i.e. a control bandwidth). The control bandwidth

11 is between the off state (no power to the electromagnet;

12 fluid not activated) and the on state (electromagnet

13 fully on; fluid fully activated).

14

15 It is important that off-state force is sufficiently low

16 for the control bandwidth of the MRF device to act over

17 the operating range of the product to which it is fitted.

18 Should the off-state force (required to move the MRF

19 device) be outside or near the upper limit of the

20 operating range, the control bandwidth of the MRF device

21 is of little benefit to the product to which it is

22 fitted, and a passive vibration control solution would be

23 better considered.

24

25 A low off state force capability can be achieved by:

26

27 1. Reducing the viscosity of the MR fluid, while avoiding

28   significant reduction in the % volume of carbonyl iron

29   content (that would reduce the on state capability).

30 2. Increasing the gap between the electromagnet and the

31   steel sleeve / cylinder, without increasing the gap to

32   an extent that the magnetic field strength (generated

1   with the electromagnet activated) becomes dissipated -

2   i.e., reducing the on state capability.

3   3. Specifying seals with sufficient elasticity to

4   maintain the MR fluid stays in the outer cylinder, but

5   avoids significant energy being absorbed by the seals

6   as the inner is forced to move relative to the outer.

7

8   The MRF dampers 10,110 of the present invention operate

9   with a low viscosity fluid to ensure a low off state

10 force is maintained. Concerns with settling and in-use-

11 thickening (where the activated MR fluid degrades to a

12 paste-like consistency) are significantly reduced if the

13 MR fluid in the control volume and particularly the

14 control region can be re-circulated (i.e., with MR fluid

15 not exposed to the magnetic field). Spaces are provided

16 in the chambers 29,128 on either side of the control

17 region 27,132 for this purpose.

18

19 The variable dampers of the present invention have a wide

20 range of applications, and the scope of the invention

21 should not be construed as being limited to a particular

22 application. Thus reference is now made to Figure 4 which

23 illustrates a vibration control system, generally

24 indicated by reference numeral 50, according to an

25 embodiment of the present invention.

26

27 Figure 4 illustrates a standard vibration control system

28 50, which controls movement between two moving objects 52

29 and 54. It will be appreciated that one object eg. item

30 54, may be static and that all movement is upon the first

31 item 52. Located between the objects 52, 54 is a spring

32 56. Also located between the two objects is a vibration

33 damper 210. The housing or outer cylinder 212 of the

1 damper is connected to the first object 54 while the

2 inner shaft 214 of the damper 210 is connected to the

3 second object 52.

4

5 Further, at positions where the dampers meet the object

6 there is located an accelerometer 58a,b. The

7 accelerometers 58a,b give an indication of the movement

8 each of the objects 52, 54. The output of each

9 accelerometer 58a,b spread to a control unit 60 from

10 which an electrical signal passes to a damper 210 to

11 provide the magnetic field on the magnetorheological

12 fluid within the damper 210. An optimised algorithm is

13 programmed into the control unit 60 onto a microprocessor

14 so that the electrical signal to the damper can be varied

15 in accordance with the values determined from the

16 accelerometers 58a,b.

17

18 Damper 210 is as described with reference to Figures 1

19 and 2. While the control unit 50 could use the second

20 embodiment shown in Figure 3, the single axis damper of

21 Figures 1 and 2 is most appropriate. The advantage of

22 using the active damper of Figures 1 and 2 is that it

23 provides a low off-state indicated by the

24 transmissibility against frequency curve obtained. This

25 is as indicated in Figure 5.

26

27 Reference is now made to Figure 5(a) of the drawings,

28 which indicates a theoretical graph showing

29 transmissibility against frequency. Drawn as a Bode

30 Magnitude Diagram of magnitude (dB) against frequency

31 (rad/sec), curve (a) of the graph shows the classic

32 initial peak and trailing edge normalised characteristic

33 of the mass 52 supported by the spring 56 alone and

1 having a resonant frequency f_n. In the same system with a

2 fixed damper, curve (b), the peak in transmissibility is

3 reduced to near one, however the trailing edge provides a

4 higher transmissibility at the high frequencies. The

5 active damper 210 provides an optimal curve, curve (c),

6 with a low off-state. This curve provides the smoothness

7 to the peak with a transmissibility of close to one,

8 matching that of the fixed damper. On the trailing edge,

9 it has properties close to the ideal properties of the

10 spring at the higher frequencies.

11

12 Experimental results taken from a sprung mass system

13 similar to the theoretical one in Figure 5(a) are plotted

14 in Figure 5(b) showing the characteristics of both the

15 spring alone, curve (d), and the optimally controlled

16 system, curve (e). The relative normalised

17 transmissibility improvement using the optimally

18 controlled variable damper 212 against the spring only

19 system is shown in difference graph of Figure 5(c)

20 whereby a positive figure indicates a reduction of

21 vibration. A single figure of relative merit of 0.8229

22 for this improved system is derived by the ratio of

23 integral of transmissibility from 0->100 Hz for the

24 controlled damper system over integral of

25 transmissibility for the spring only system. The equation

26 for this is given:

27

28
$$\frac{\int_{0}^{100Hz} trans(ControlledDamper)}{\int_{0}^{100Hz} trans(SpringOnly)}$$

29

30 Any comparison to alternative damper designs should be

31 evaluated using the baseline of the spring only graph

1 given in Figure 5(b) in order that a fair comparison can

2 be made. This means that a system with a replica

3 transmissibility function would need to be used.

4

5 It is estimated that with ongoing refinements to the

6 system to improve optimisation, this relative figure of

7 merit can be brought down to 0.5 or better.

8

9 The control system of Figure 4 finds particular

10 application in respect of engine mounts, pump mounts and

11 power tools. The engine block could be mounted at the

12 object marked 52 and thus provides the vibration control

13 for pumps, generators, engines and in vehicle

14 manufacture.

15

16 A further embodiment of a vibration control system,

17 generally indicated by reference numeral 150, is

18 illustrated in Figure 6. Like parts to those have been

19 given the same reference numeral with the addition of

20 100. The system 150 now incorporates a pivot 53. In

21 this way the load, or first object 152 acts on the damper

22 212 against the pivot 53. The system 150 may incorporate

23 accelerometers and a control unit as described with

24 reference to Figure 4.

25

26 The spring 156 in this embodiment is formed by leaf

27 springs as distinct from the typical coil spring. These

28 leaf springs can be advantageously arranged around the

29 damper 212 to save space and provide better control to

30 the system 150. A non-exhaustive selection of suitable

31 arrangements is illustrated in Figures 7(a)-(d). Figure

32 1(a) illustrates a crossed pair over the damper 212. Each

33 spring 156 is in a 'c' spring formation with both ends on

1 the second object 154. Figure 7(b) illustrates a triplet

2 arrangement of three springs, meeting at the first object

3 152 and distributed across the second object 154. Figure

4 7(c) provides a quad arrangement of four independent leaf

5 springs 156(a)-(d) which are symmetrically distributed

6 around the damper 212. Figure 7(d) illustrates an

7 arrangement for adjusting the resonant frequency of the

8 spring 156 by layering individual leaf springs, creating

9 a progressive type leaf spring. The lowest spring in the

10 configuration is in a 'c' spring formation with the

11 additional springs symmetrically aligned on its upper

12 surface.

13

14 As a particular example, the invention will now be

15 described as is incorporated in a snowboard.

16

17 This application is shown generally in Figure 8. A board

18 340 has bindings 342 with shim portions 344, to which the

19 outer portion 312 of a damper (or "actuator") 310 is

20 attached. The inner portion 314 of the damper 310 is

21 attached to the board 340. Torsion forks 346 are also

22 mounted on the board 340, and are also in communication

23 with the inner portion 314 of the damper 310. The damper

24 310 primarily as described herein with reference to

25 Figure 3.

26

27 As is described in more detail below, sensors monitor

28 dynamic movement and provide input to an intelligent

29 control unit (ICU) made up of one or more

30 microprocessors. The response (i.e., energy absorbing

31 capability) of the MRF actuator(s) controls dynamic

32 movement of the product with a view to optimising

1 performance/tuning the system to suit the operator
2 player).
3
4 The multi-axis damper 310 aims to provide a wide
5 bandwidth of semi-active damping. The system will enable
6 the level of vibration energy absorption to be adapted
7 with respect to vibration impulses (i.e., the product of
8 force and time) and can be tuned to suit the user.
9
10 Soft flex, torsionally flexible boards are easier to
11 turn and better to control at lower speeds and are
12 generally better off piste. Stiff, torsionally rigid
13 boards have greater stability at speeds and have enhanced
14 carving ability - making it easier to place the board in
15 a turn at speed. Damper 310 is capable of adapting the
16 characteristics of the board with respect to speed and
17 snow conditions. This is achievable by using integrated
18 sensors to monitor the amplitude and time response of
19 vibrations that can be used to characterise speed and
20 surface condition, with an algorithm programmed into a
21 microprocessor controlling power supply to the
22 electromagnets that adapt the energy absorbing capability
23 of the MRF actuator(s)ie. dampers 310.
24
25 The actuator must be mounted so that torsional and
26 longitudinal movement of the board can be transmitted
27 through the actuator.
28
29 For a snowboard, the actuator can be mounted with its
30 central axis 334 either transverse or parallel ("in-
31 line") to the longitudinal axis of the board 340.
32

1 Figure 9 shows a third embodiment of the present

2 invention, namely an actuator 350 that is mounted in line

3 with the board 340. Components of the actuator 360 are

4 similar to the components referred to in Figure 3 and

5 shall not be hereinafter described in detail. The

6 reference numerals that apply to Figure 3 can be taken to

7 refer to the corresponding components in Figure 9 now

8 prefixed '3'.

9

10 The sprung collar or the bush described with reference to

11 Figure 3, are not essential parts of the MRF device when

12 it is incorporated in to a snowboard, as the board acts

13 as the spring that is to be controlled. MRF devices with

14 sprung collars or bushes would add to the stiffness

15 matrix of the board and provide adaptive semi-active

16 control of dynamic movement. On the snowboard, the

17 actuator is returned to a neutral position as the board

18 relaxes after being deflected (assuming the board does

19 not become permanently deformed).

20

21 The electromagnet 320 is powered by power supply 352

22 routed through the shim 344. The MRF chamber 328 is

23 attached to the board 340, and the outer portion 312 of

24 the actuator 360 is attached to the shim 344. If the

25 actuator 360 is transversely mounted, the chamber 328 and

26 outer potion 312 are connected to the board 340 and shim

27 344 also.

28

29 Steel sleeve 354 is attached to the outer cylinder 312 of

30 the outer member, and has the shape of a cylindrical body

31 portion with two washer shaped end portions at each end

32 of the cylinder, the outer edges of which are in line

33 with the outside perimeter of the body portion. The

1   electromagnet 320 is mounted on an axle and positioned

2   inside the steel cylinder 354.

3

4   The inner axle and outer cylinder share a common axis.

5   There is a defined gap between the electromagnet and the

6   steel cylinder, comprising a first dimension X, being the

7   distance between the end of the electromagnet 320 and the

8   inner wall of the steel cylinder 354, and a second

9   dimension Y, being the distance between the inside

10   diameter of the steel cylinder 354 and the outside

11   diameter of the electromagnet 320.

12

13   The gaps as defined by the dimensions X and Y enable

14   the device to control up to six degrees of freedom.

15

16   To minimise the off state force, X and Y should be made

17   as large as possible, bearing in mind that their increase

18   will result in a corresponding decrease in the force that

19   can be provided by the device once the full on state is

20   applied.

21

22   As shown in Figure 8, the torsion forks 346 are connected

23   to the inner portion 314, to transmit longitudinal and

24   torsional movement to the MRF actuator.

25

26   The MRF actuator 360 can adapt semi-active damping of

27   torsional and longitudinal movement with a combination of

28   the pressure driven flow (/ valve mode) and direct shear

29   mode of the MR fluid being applied.

30

31   Mounted with its axis parallel to the axis of the board,

32   the MRF actuator can adapt semi-active damping of

33   longitudinal movement with a combination of the pressure

driven flow (/valve mode) and direct shear mode of the MR
fluid being applied. Torsional stiffness can be adapted
by applying the direct shear mode to resist rotation of
the inner relative to the outer.

Adaptive control of the damping is provided by an
intelligent control system. Figure 10 shows an
intelligent control system 90 suitable for use with the
damper 360 shown in Figure 9. It will be appreciated that
a similar control system would be suitable for a
transversely mounted actuator.

Integration of a parasitic power generator is preferable
to powering the system from a battery. A piezo-ceramic
power generator 70 (such as PZT - lead zirconium
titanate) located at areas of concentrated load can be
used to harvest power from deflections induced by the
movement between the rider and the board.

The location of the generator could for example, be
specified to be under the riders boot. For example, the
power generator could be a piezoelectric (lead-zirconium
titanate - PZT) bimorph / piezoelectric (PZT) unimorph
located in the minding foot-plate/ between the binding
assembly and the deck of the board.

This is in contrast to presently available systems, which
merely use the vibration caused by movement of the board
to generate power. By placing the piezo-ceramic power
generators 70 at strategic points where there is
concentrated load and/or movement from the rider of the
board when using it, enough power can be generated to
power the electromagnet and ICU.

1 The piezo-ceramic generator 70 located within the binding

2 assembly (/between the binding and board) can power an

3 energy efficient network of control-actuator(s).

4

5 An array of piezo (polymer) sensors (e.g.,

6 polyvinylidenefluoride - PVDF) sensors 72 provides a

7 self-powered vibration monitoring capability. An array of

8 sensors 72 located within the beam section to be

9 controlled can provide input to the control interface on

10 longitudinal and torsional dynamic movement produced from

11 surface induced impulses.

12

13 This system must be sufficiently energy efficient so that

14 the power available to the electromagnet can sufficiently

15 change the apparent viscosity of the MR fluid, resulting

16 in a satisfactory improvement in dynamic control.

17 Therefore, the number of turns on the core of the

18 electromagnet must be sufficient to generate a

19 satisfactory on state. but be conservative in number to

20 conform to the power constraints. The available energy

21 and required control bandwidth must be considered for

22 each application.

23

24 The data provided by the sensors 72 can be used to

25 determine the amplitude and frequency characteristics of

26 board vibration induced as the board 340 moves over the

27 snow. Characteristics of the vibration can be used to

28 determine environmental inputs (e.g., hard / soft packed

29 snow), based on information pre-programmed into the lCU

30 90.

31

32 The ICU 90 controls the power supply to the electromagnet

33 20 such that vibration amplitude and frequency may be

1 controlled subject to the applied control algorithm

2 (e.g., proportional control/proportional-integral-

3 differential control/sky-hook algorithm/to a set value -

4 up to a definable maximum).

5

6 One or more MRF actuators may be mounted transversely, or

7 with its axis parallel to that of the board as described

8 above in order to provide multi-axis control.

9

10 Another major application of the present invention is the

11 incorporation of an actuator in the grip of sports

12 equipment, such as for example tennis, squash or

13 badminton rackets; golf clubs; baseball or cricket bats;

14 or polo mallets.

15

16 Figure 11 shows the application of an adaptive shock

17 absorbing grips may integrated on a golf club 80.

18

19 The MRF device 110 is integrated so that the axis is in-

20 line with the axis of the shaft 82, with the inner

21 component mounted to the shaft 82 and the outer making up

22 the grip. Activation of the electromagnet mounted on the

23 structural inner component results in an apparent

24 viscosity change in the MR fluid between the inner and

25 outer (grip), reducing relative movement in two axes and

26 introducing an adaptable energy absorbing capability.

27

28 A spring return to a neutral position is required. Sprung

29 collars or bushes can be located between the seals

30 against the inner axle and outer cylinder (i.e. in the

31 space between the shaft and the outer housing) to provide

32 resistance to deflection that the MR fluid is able to

1 dynamically control. Therefore the spring is integrated

2 in the damper assembly.

3

4 A contact plate can interface the shock-absorbing grip

5 with the sensor-control and power supply elements of the

6 system.

7

8 For this application, and application to handles of other

9 devices, it is desirable to actively prevent translation

10 along and rotation about the shared central axis of the

11 inner member relative to the outer member, so the off

12 state force in these degrees of freedom needs to be

13 raised.

14

15 This is possible by specifying seals with appropriate

16 elasticity to prevent noticeable movement.

17

18 Again, integration of a parasitic power generator is

19 preferable to powering the system from a battery. A

20 piezo-ceramic power generator located at a point of

21 concentrated load can be used to harvest power from

22 deflections induced by the movement between the head or

23 club, the shaft, and the handle (where the grip is

24 located). The piezo-ceramic generator can power an energy

25 efficient network of control-actuator(s), with piezo

26 (polymer) sensors providing self-powered vibration

27 monitoring capability.

28

29 PVDF sensors are proposed to provide a self-powered

30 vibration monitoring capability. An array of sensors

31 located within the shaft can provide input to the control

32 interface on transmitted vibrations resulting from shock

33 induced impulses.

1   A further identified application of multi-axis adaptive

2   semi-active control of dynamic movement is in bicycle and

3   motorbike handles. Sports bikes with low handles result

4   in a riding position that puts weight on the rider's

5   wrists, with fatigue compounded by any shock induced

6   vibration that is not sufficiently damped by the main

7   front suspension. One or more multi-axis MRF device can

8   be located in the bike handles as a secondary system to

9   absorb shock and reduce wrist fatigue.

10

11   In a motorcycle application there is sufficient capacity

12   to power the MRF device(s) with negligible performance

13   consequences.

14

15   Applied to bicycles, although it is possible, it is

16   advantageous not to power the MRF device(s) from the

17   powertrain (i.e., rotation of the pedals / the wheels) as

18   this will reduce performance. An alternative, to a dynamo

19   powering the MRF device(s) from the powertrain, is a

20   parasitic power generator - preferably located between

21   the bicycle and rider, at a position, where there is a

22   concentrated load.

23

24   A piezo-ceramic power generator located in the seat-post

25   can be used to harvest power from deflections induced by

26   the movement of the rider on the seat. The piezo-ceramic

27   generator can power an energy efficient network of

28   control-actuator(s), with piezo (polymer) sensors

29   providing self-powered vibration monitoring capability.

30

31   The single axis damper with additional supports provides

32   a damper with the lowest possible off-state. When

33   incorporated in a vibration control system, as could be

1 used on engines, pumps, generators etc, the damper

2 provides a system with an ideal transmissibility against

3 frequency curve.

4

5 Improvements and modifications can be made to the above

6 without departing from the scope of the present

7 invention. In particular, the application of the

8 invention to be incorporated in specific devices is not

9 limited to the list of specific devices herein.

10 Furthermore, it will be apparent that the specific

11 geometry of, for example, the layout of the sensor array

12 or of the parasitic power generators may be varied as

13 appropriate for the specific application being

14 considered.

15